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DESIGN OF PHASE-ANGLED BALANCE WEIGHTS FOR AN INVERTER DRIVEN SCROLL COMPRESSOR

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ABSTRACT

This paper presents a new design method of balance weights with phase angles for scroll compressors. Based on force and moment balances on crankshaft and compressor frame, mathematical formulation of shaft bearing loads, unbalanced force and moment acting on the frame has been made in terms of balance weight design parameters such as masses and phase angles of the upper and lower balance weights. Any two of crankshaft main bearing load, sub bearing load, unbalanced body force, and overturning moment can be controlled by the balance weight design parameters.

NOMENCLATURE

F_3, F_4	Reactions between oldham ring keys and frame
F_a, F_{rg}, F_{tg}	Gas forces in axial, radial, and tangential directions, respectively.
F_{Body}	Unbalanced body force or resultant force acting on the frame
F_{cp}	Total resultant force acting on crank pin
F_{cpc}, F_{csc}, F_{orc}, F_{osc}, F_{sbc}	Centrifugal forces of crank pin, crankshaft, oldham ring, orbiting scroll, and slider bush, respectively
F_{dw}, F_{uw}	Centrifugal forces of lower and upper balance weight, respectively.
F_{mj}, F_{sj}	Main and sub bearing loads at crankshaft, respectively.
F_{rs}	Radial sealing force between wraps of fixed and orbiting scroll members
l_{cp}, l_{cs}, l_{dw}, l_h, l_{sj}, l_{uw}	Lengths defined in Fig. 2 and Fig. 3
M_{Body}	Overturning moment or resultant moment acting on the frame
m_{dw}, m_{uw}	Masses of lower and upper balance weights, respectively
r_d, r_u	Radii of mass center of lower and upper balance weights, respectively
r_{cp}, r_s	Eccentricity of crank pin, and orbiting radius, respectively
r_x, r_y	Radial and tangential reaction positions at thrust surface
β_1, β_2	Phase angles of upper and lower balance weights
ω, θ	Angula velocity of crankshaft and crank angle, respectively

Subscripts

c, n	Conventional and new, respectively
r, t	Radial and tangential components, respectively.

INTRODUCTION

Conventional balance weight design for scroll compressors typically employs two balance weights: upper one is circumferentially positioned 180 deg. from the crank pin and lower one on the same side of the pin. Sizing of the balance weights is made to satisfy the balance of centrifugal forces of moving members and the moment balance due to these forces. Nieter and DeBlois[1] suggested an optional method of designing balance weights which theoretically eliminates reaction forces at shaft bearings by giving a phase angle to upper balance weight, keeping lower one 180 deg. out of phase to the upper one. As they pointed out, however, the difference in the centrifugal force of balance weights defined by their method and those defined by the conventional method would directly go into increasing the frame vibration.

In the present paper, we aim to show that, while unbalanced body force on the compressor body is being kept minimal, one of shaft bearing loads or overturning moment of the compressor frame can be controlled to desired values by providing the balance weights with individual phase angles.

ANALYSIS OF FORCE AND MOMENT BALANCES ON SHAFT AND FRAME

Fig.1 shows a schematic of the phase-angled balance weights considered in this study. The individual force components of the balance weights are as follows: $F_{uwr}=m_{uw}r_u\omega^2\cos\beta_1$, $F_{uwt}=m_{uw}r_u\omega^2\sin\beta_1$, $F_{dwr}=m_{dw}r_d\omega^2\cos\beta_2$, $F_{dwt}=m_{dw}r_d\omega^2\sin\beta_2$. Axial positions of the balance weights are fixed as shown in Fig. 2. Hence, the balance weight design is composed of finding four independent variables, m_{uw}, m_{dw}, β_1, and β_2, whose combination would give desired values of shaft bearing loads, unbalanced body force, and overturning moment.

Various forces and their acting points on the crankshaft are shown in Fig. 2. Shaft bearing loads can be calculated from force and moment balances on the crankshaft as follows.

$$F_{sjr}=\frac{1}{l_{sj}}[l_{cp}(F_{cpr}+F_{cpc})+l_{uw}F_{uwr}+l_{cs}F_{csc}-l_{dw}F_{dwr}] \tag{1}$$

$$F_{sjt}=\frac{1}{l_{sj}}[l_{cp}F_{cpt}+l_{uw}F_{uwt}-l_{dw}F_{dwt}] \tag{2}$$

$$F_{mjr}=\frac{1}{l_{sj}}[(l_{cp}+l_{sj})(F_{cpr}+F_{cpc})-(l_{sj}-l_{cs})F_{csc}-(l_{sj}-l_{uw})F_{uwr}+(l_{sj}-l_{dw})F_{dwr}] \tag{3}$$

$$F_{mjt}=\frac{1}{l_{sj}}[(l_{cp}+l_{sj})F_{cpt}-(l_{sj}-l_{uw})F_{uwt}+(l_{sj}-l_{dw})F_{dwt}] \tag{4}$$

Fig. 3 shows reactions from various members to the frame. F_{rg}, F_{rs}, and F_{tg} are transmitted from orbiting scroll to the frame via fixed scroll which is rigidly attached to the frame, and F_a is to the thrust surface. Resultant of these forces is the unbalanced force acting on the frame, F_{Body}. By using relations of reaction forces of various members[2], radial and tangential components of F_{Body} are obtained by the equations (5) and (6), respectively.

$$F_{rB}=F_{osc}+F_{sbc}+F_{cpc}-F_{csc}+F_{orc}\sin^2\theta-F_{uwr}+F_{dwr} \tag{5}$$

$$F_{tB}=F_{orc}\sin\theta\cos\theta+F_{uwt}-F_{dwt} \tag{6}$$

With the moment center at the sub bearing position, components of overturning moment acting on the frame are obtained by the equations (7) and (8).

$$M_{tB}=-(l_{sj}+l_{cp}+l_h)(F_{rs}+F_{rg})-(r_s-r_x)F_a-l_{sj}F_{mjr} \tag{7}$$

$$M_{rB}=-(l_{sj}+l_{cp}+l_h)F_{tg}+r_yF_a+l_{sj}F_{mjt} \tag{8}$$

While we have eight equations (1)-(8), there are only four unknowns: F_{uwr}, F_{uwt}, F_{dwr}, and F_{dwt}. This means that not all of the eight items (F_{mjt}, F_{mjr}, F_{sjt}, F_{sjr}, F_{tB}, F_{rB}, M_{tB}, and M_{rB}) can be set to independent arbitrary values. Instead, only four of them can be given independent values, and the remaining four items should depend on the others.

In the following example calculation, two components of the unbalanced body force, F_{tB} and F_{rB}, and two components of the shaft main bearing force, F_{mjt} and F_{mjr}, will be selected as independent ones. In particular, if F_{tB} and F_{rB} are set to zero, the equations (5) and (6) give the following relations.

$$F_{dwr} = F_{uwr} - (F_{osc} + F_{sbc} + F_{cpc} - F_{csc} + F_{orc}/2) \tag{9}$$

$$F_{dwt} = F_{uwt} \tag{10}$$

Since the terms including θ, however, can not be taken into consideration in the equations (9) and (10), F_{Body} itself can not be made real zero. By substitution of the equations (9) and(10) into (5) and (6), minimum of F_{Body} can be found as follows: $F_{Body} = \sqrt{F_{tB}^2 + F_{rB}^2} = 1/2 F_{orc}$. Also, the equations (3) and (4) can be rewritten for F_{uwr} and F_{uwt} by using the equations (9) and (10).

$$\begin{aligned} F_{uwr} = \frac{1}{l_{dw} - l_{uw}} [& (l_{sj} + l_{cp}) F_{cpr} + (l_{cp} + l_{dw}) F_{cpc} - (l_{dw} - l_{cs}) F_{csc} \\ & - (l_{sj} - l_{dw})(F_{osc} + F_{sbc} + F_{orc}/2) - l_{sj} F_{mjr}] \end{aligned} \tag{11}$$

$$F_{uwt} = \frac{1}{l_{dw} - l_{uw}} [(l_{cp} + l_{sj}) F_{cpt} - l_{sj} F_{mjt}] \tag{12}$$

From these four equations (9)-(12), four unknowns of F_{uwr}, F_{uwt}, F_{dwr}, F_{dwt} can be calculated for given F_{mjt} and F_{mjr}. And, in turn, once F_{uwr}, F_{uwt}, F_{dwr}, F_{dwt} are found, the design parameters, m_{uw}, m_{dw}, β_1, and β_2 can be determined.

CALCULATION RESULTS AND DISCUSSIONS

The present method of phase-angled balance weights has been applied to a radially compliant 3 hp class scroll compressor. Instead of using individual main bearing load components F_{mjt} and F_{mjr} as independent inputs for the balance weight design, the resultant load F_{mj} together with the phase angle of the upper balance weight, β_1 are used for practicality. An example of correlation between these two pairs is shown in Fig. 4(a), where variation of β_1 controls the distribution between F_{mjt} and F_{mjr} for given F_{mj}. As β_1 is varied from 0° to 360° for predetermined F_{mj} and minimal F_{Body}, the other design parameters, β_2 and m_{uw} and m_{dw} vary as in Fig. 4(b) and (c), respectively. And the corresponding F_{sj} and M_{Body} are calculated as shown in Fig. 5(a) and (b), respectively. Minima of both F_{sj} and M_{Body} take place at about the same β_1, the locations of which are marked with diamond marks in the figures. Fig. 6 shows that decrease in F_{mj} is accompanied by increase in M_{Body}, and that there is an optimum value of F_{mj} for minimum F_{sj}.

Shaft bearing loads, unbalanced body force, and overturning moment of the present method and those of the conventional method are compared at various compressor speeds in Fig. 7(a)(b)(c), respectively. In the figures, 'A' and 'B' represent the conditions specified in Fig. 6, and 'C' stands for the conventional method. The trend of monotonic increase in all of the items with increasing speed is general, except F_{sj} of the condition B. For F_{Body}, the conventional method gives a little larger value. Furthermore, as shown in Fig. 8, considerable variation in F_{Body} with the crank angle exists for the conventional method, while no such variation for the present method. Fairly steadiness in F_{Body} of the present method results from inclusion of the centrifugal force of oldham ring in the radial force balance on the compressor frame.

CONCLUSIONS

From the application of phase-angled balance weights to a radially compliant scroll compressor, the following conclusions can be drawn:

1. By circumferential positioning of balance weights in addition to sizing, any two of the following items can be controlled to desired values: shaft main bearing load, sub bearing load, unbalanced body force, and overturning moment of the body.
2. With minimum unbalanced body force, reduction in the shaft main bearing load can be obtained at the expense of increase in the overturning moment.
3. The unbalanced body force can not be totally eliminated, mainly because the centrifugal force of the oldham ring changes with the crank angle. Consideration of its mean value into the radial body force balance results in the removal of variation in the unbalanced body force with the crank angle.

REFERENCES

[1] Nieter, J. J. and DeBlois, R. L., "Counterweighting scroll compressor for minimal bearing loads," Intern. Compr. Eng. Conf. at Purdue, 1988, pp.175-181

[2] Kim, H. J. and Kim, J. H., "Balance weight design for a radially compliant scroll compressor," Intern. Compr. Tech. Conf., Chengdu, 1997, pp.232-239

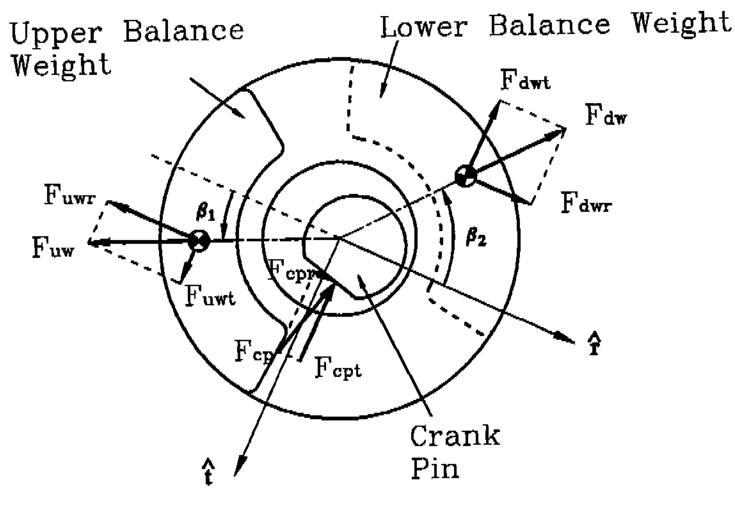

Fig. 1 Phase-angled balance weights and their force components

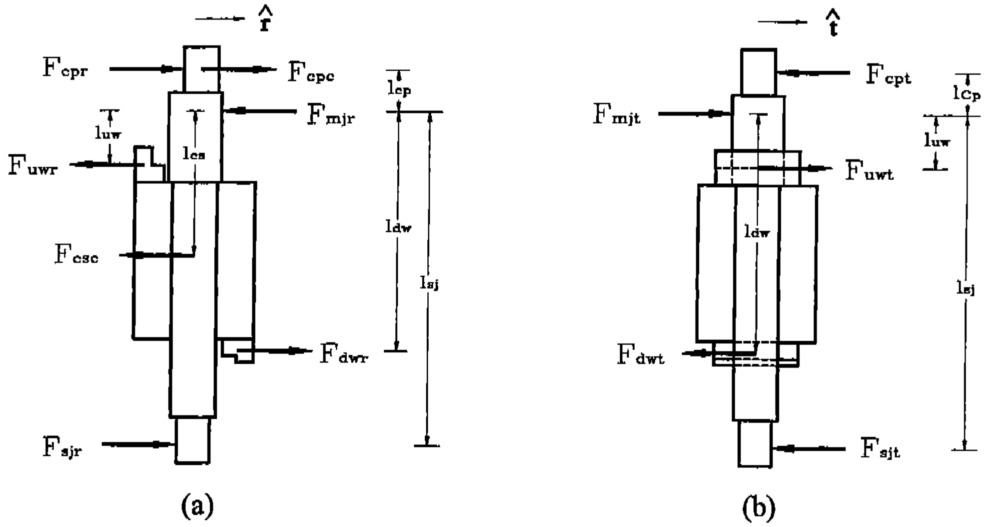

Fig. 2 Forces acting on crankshaft. (a) Radial forces; (b) tangential forces

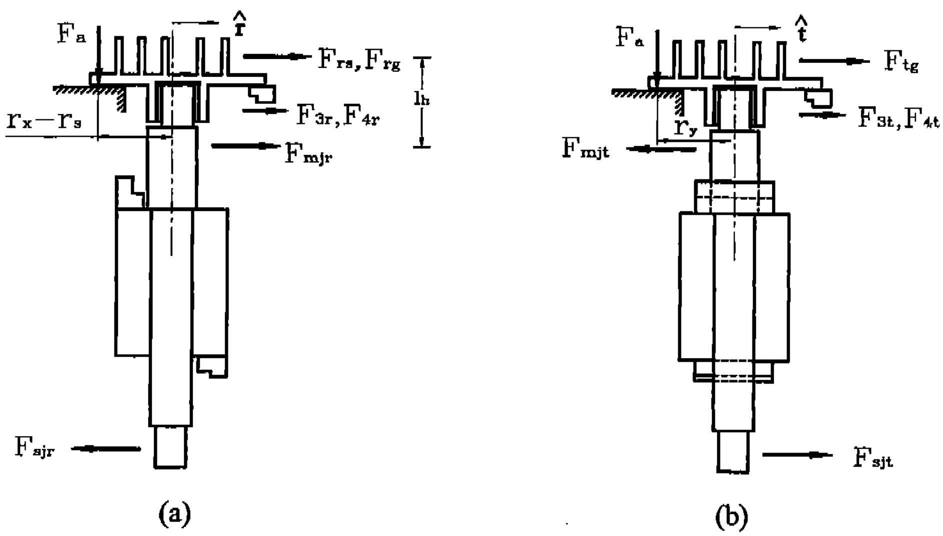

Fig. 3 Forces acting on compressor frame. (a) Radial forces; (b) tangential forces

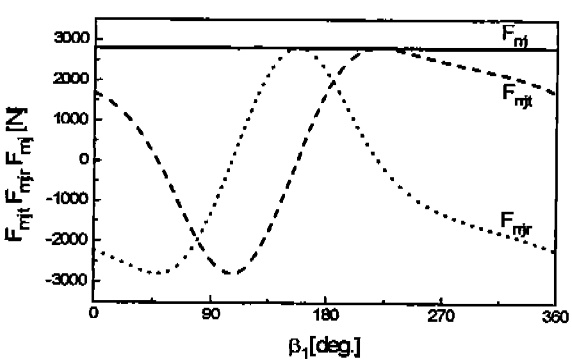

Fig. 4(a) Effects of upper balance weight phase angle on main bearing load components.

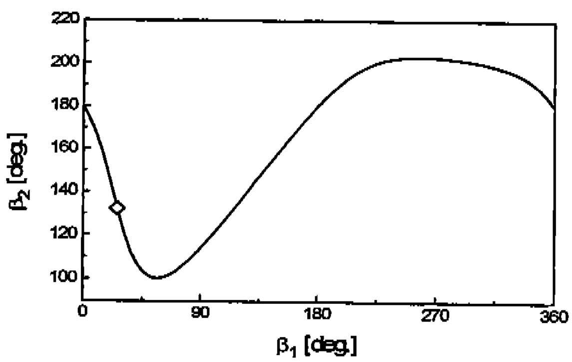

Fig. 4(b) Effects of upper balance weight phase angle on lower balance weight phase angle.

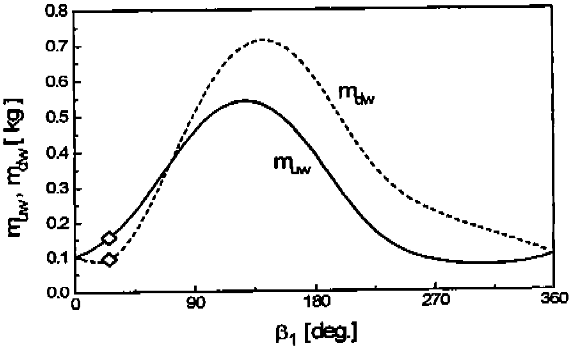

Fig. 4(c) Effects of upper balance weight phase angle on masses of balance weights.

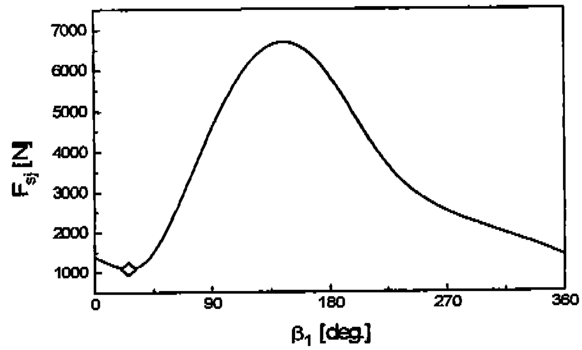

Fig. 5(a) Effects of upper balance weight phase angle on sub bearing load at F_{mj}=2800N.

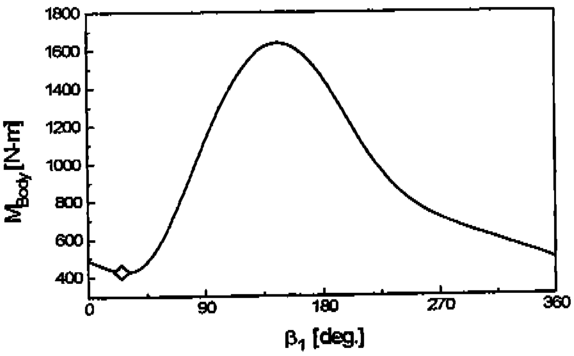

Fig. 5(b) Effects of upper balance weight phase angle on overturning moment at F_{mj}=2800N.

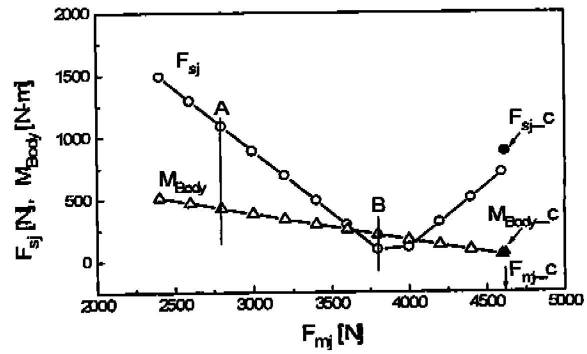

Fig. 6 Variations of sub bearing load and overturning moment with main bearing load.

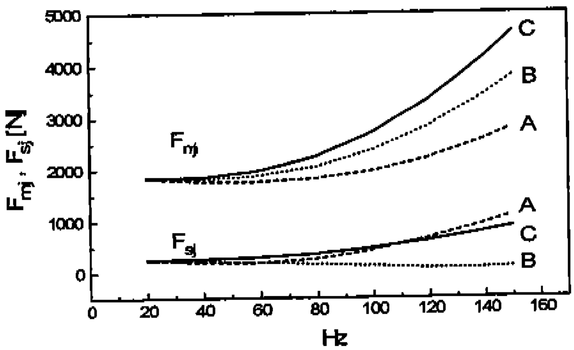

Fig. 7(a) Effects of compressor speed on balance weight performance : shaft bearing loads.

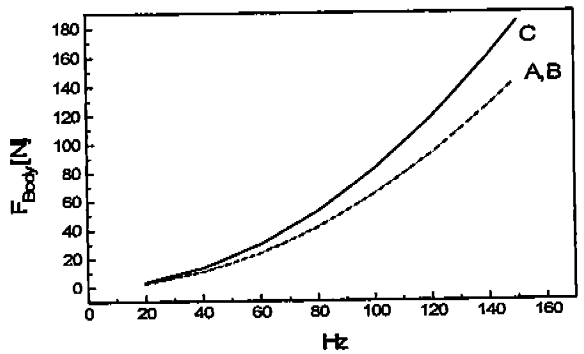

Fig. 7(b) Effects of compressor speed on balance weight performance : unbalanced body force.

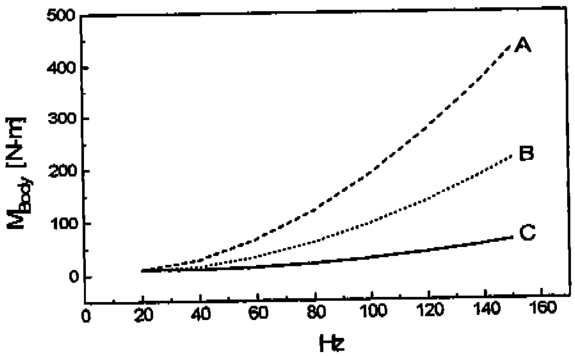

Fig. 7(c) Effects of compressor speed on balance weight performance : overturning moment.

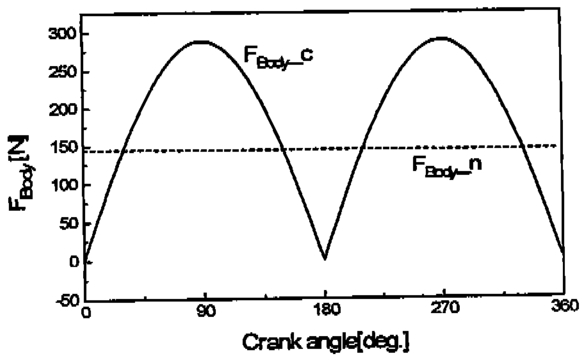

Fig. 8 Variation of unbalanced body force with crank angle.